EXPERIMENTAL STUDIES OF TWIST RATIO EFFECT TO THE HEAT TRANSFER ENHANCEMENT USING SQUARE CUT TAPE AND CLASSICAL TAPE INSERT

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Keywords:

Square cut tape insert
Friction factor
Heat exchanger
Nusselt number
Twist ratio

Abstract:

This research was conducted experimentally to examine the effect twist ratio on heat transfer enhancement in the concentric tube heat exchanger with the addition of square cut and classical cut tape insert. Test section was the single pass concentric tube heat exchanger with inner tube and outer tube made of aluminum. Flows in the inner tube and in annulus were counter flow. Working fluid in the inner tube was hot water which its inlet temperature was maintained at 60°C, whereas the working fluid in the annulus was cold water. square cut and classical cut tape insert tape insert with the twist ratio (Y) variation = 2,7; 4,5; and 6,5 installed in the inner tube of the concentric tube heat exchanger. Research's result shown that heat transfer enhancement in the concentric pipe with augmentation at the same Reynolds number, the addition of square cut and classical cut tape insert with the twist ratio (Y) = 2,7; 4,5; and 6,5 into the inner tube increase the Nusselt numbers has been 79%; 58,2%; 43,5% and 67,8%; 48,7%, 31,1% compared to the plain tube, respectively. At the same Reynolds number, the addition of square cut and classical cut tape insert with the twist ratio (Y) = = 2,7; 4,5; and 6,5 into inner tube has produced an friction factor 2,58%; 2,2%; 1,74% dan 2,41%; 1,96%, 1,44% than the friction factor of plain tube, respectively. The addition of square cut and classical cut tape insert with the twist ratio (Y) = 2,7; 4,5; and 6,5 into the inner tube has produced an enhancement heat transfer ratio of 1,3; 1,2; 1,1; and 1,23; 1,1; 1,1, respectively.

INTRODUCTION

Heat transfer enhancement techniques, especially in heat exchanger, can substantially improve its performance. The general objective of this technique is to reduce the heat exchanger size, increase its capacity, and cut down heat exchanger pumping power. It can be classified into three groups; techniques of passive, active, and mix. Its passively technique is acquired without providing additional energy flow. The active one is conducted by providing extra energy to the fluid flow, so that the active technique requires higher costs than passive techniques. In a mix of techniques, two or more of the active and passive techniques are used simultaneously to generate heat transfer enhancement, where it is higher than other techniques which are operated separately.

Twisted tape inserts is one of the techniques which practiced to boost a passively heat transfer on heat exchanger. It became most popular due to their low cost and easy installation. On the other hand, twisted tape inserts which purposed as a continuous flow twisting device to enhance the heat transfer rate.

Research on heat transfer enhancement of heat exchanger has ever performed by employing inserts with v-cut punched and twisted tapes. Test section has 700 mm of display dimension, 26 mm of inner diameter, 30 mm of outside diameter, 2 mm of thickness. Punched and v-cut twisted tapes operated variations in 9, 10, and 11 of twist ratio. The results portrayed that its heat transfer rising in comparison with the plaint tube from 3.34 to 14.4% for 9 of twist ratio and 13.35 to 25% with a twist ratio of 11. Its maximum friction factor was 52% at 9 of twist ratio and 66% at 11 of twisted ratio that compared with plain tube. (Quazi and Mohite-2015).

Previously, also conducted an experimental research on heat transfer characteristics, friction factor, and thermal performance of turbulent flow in a round pipe with a rectangular-cut twisted tape insert. Twisted tape was made of stainless steel with 2 mm of thickness and 20 mm of width, 105 mm of length so the twist pitch ratio was 5.25. The results displayed that at the same Reynolds number, Nusselt number in rectangular-cut pipe with twisted tape inserts roses 2.3 to 2.9 compared with plain tube, with the average expansion of 2.6 times. The pipe friction factor for a rectangular-cut twisted tape insert were 39% to 80% higher than the plain tube friction factor. The thermal performance ranges were from 1.9 to 2.3. (Salam, et al 2013).

The experimental study on heat transfer characteristics of a heat exchanger with Elliptical-cut twisted tape insert on twist ratio (y= 8.0) and five ratio of major to minor (Z) were 5; 4; 3.3; 3, and 2.5.

The Reynolds number variations of 10000-19000, 14-22kw/ m^2 of heat flux variations on a plain tube and 23-40 kW/ m2 for inserted pipes. The results exhibited that the Nusselt numbers average was rised with the increassing number of Z = 5; 4; 3.3; 3, and 2.5 at 19.3%; 41.8%; 53.83%; 68.5%; 73.16%; and 84.5% respectively, it were higher compared to the plain tube. The performance of thermally-cut elliptical twisted tapes ranges from 0.91 to 1.25 for Z = 5.0; 4.0; 3.3; 3 and 2.5. (Ganorkar, et.al-2015)

This study was conducted to examine the effect of the Reynolds number variations and the effect of adding squere-cut twisted tape insert in the pipe in the (inner tube) on the heat exchanger pipe concentric annular channel on the characteristics of heat transfer and friction factor. It is expected with the addition of inserts squere-cut twisted tape inserts and twisted tape insert with a classical twist variation ratio can improve convection heat transfer coefficient of heat exchanger pipe in pipe concentric with the increase in pressure drop is acceptable.

RESEARCH METHODOLOGY

Testing Equipment and Research Procedure

Research equipments consist of three systems, such as, the hot water flow system tracks in the pipes, measurement, and path of the cold water flow in the annulus. The electric water heater with a total power of 4,000 watts was employed to heat the water in the hot water tank. On it, the hot water temperature was setup using thermocontroller to keep constant at 60°C to the hot water pipes temperature enter through inside of the concentric pipe heat exchanger. The water heat pumps for pumping hot water from the hot water tank, passing through heat exchanger test equipment and then hot water returned to the hot water tank. The testing equipment scheme can be seen in Figure 1. The fluid flow direction on the pipe and the annulus were in opposite directions.

Bypass valve was used to regulate water flow variations in the amount of heat water which entering the pipe and it values can be read with a rotameter. Cold water which flowing into the annular was constantly maintained during the test. It flowing by gravity method was the cold water flow which coming from the cold water tank that located above the cold water surface elevation in the cold water tank to keep constant by employing the overflow pipe. The cold water that comes out of the test equipment heat exchanger immediately discarded.

The U shape Manometer was employed to measure the pressure difference on the hot water which flowing at a side entrance and the exit. Water was used inside manometer. Trapped water was used to store the water-borne upon the manometer pressure measurement in order not to get into the manometer.

The temperature measurement of the cold water at in and out annular, outer wall temperature and the hot water temperature in the pipe at in and out was using K type thermocouple temperature measurement of the outer walls of the pipes which totaling 10 points that measured alternately. Thermocouple reader was exploited to read the thermocouple. The U pipe manometer fluid with water was used in measurement of the pressure drop in the pipeline. A concentric heat exchanger pipe with a one pass pipe was made of aluminum. The dimensions of the concentric pipe heat exchanger can be seen in Figure 2.

Square cut twisted tape inserts and classical tape inserts nomenclature in a pipe can be seen in Figure 3. In Figure 3, W is thick, y is the pitch square cut twisted tape insert, d is the cutting height, W is the width (twist). In the previous research the inside part of pipe was made of aluminum with 25 mm of inner diameter and 2000 mm of length. Twisted tape inserts made of aluminum with 1.5 mm of thickness and 23.5 mm of width. Plain twisted tape insert and a square-cut twisted tape which used has a twist ratio of 2.0; 4.4; and 6.0 (Murugesan, et al., 2010).

A flow Reynolds number of water in the pipe was varied by adjusting the flow rate of 2-6 LPM to pipe in without STT (plain tube) and inner pipes with STT. Data was obtained by the inlet and outlet water temperature of annulus, outer wall temperature, the water mass flow rate, and the annulus pressure drop in the pipeline. For each test variation data was collected every 10 minutes to obtain a steady state. The data of these steady state conditions were used in the research data computation and analysis. For comparison, there was performed testing on the pipeline without STT (plain tube) and with the square-cut tape inserts addition and classical twisted tape insert.

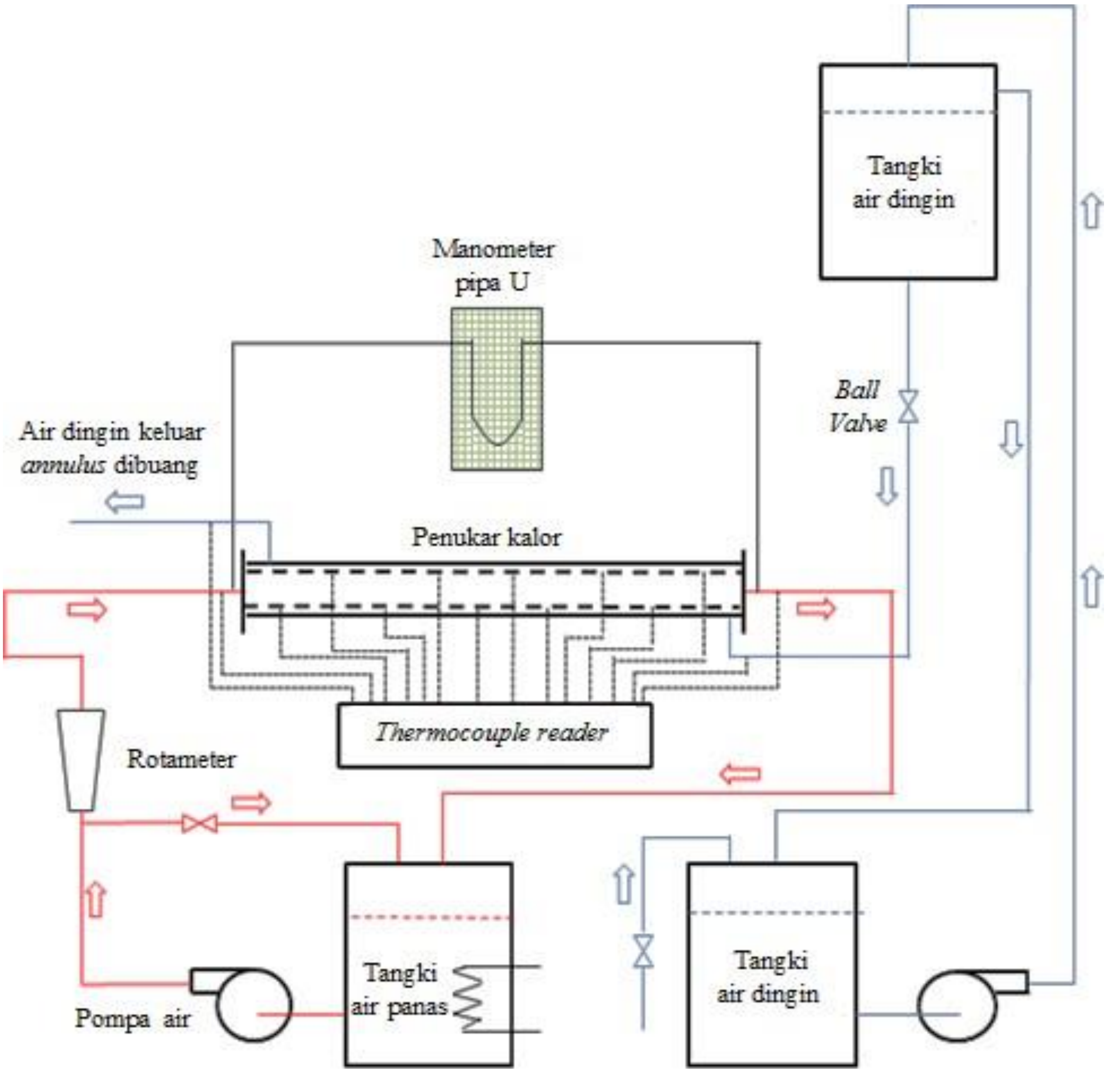

Figure 1. Concentric pipe heat exchanger scheme by square-cut twisted tape insert.

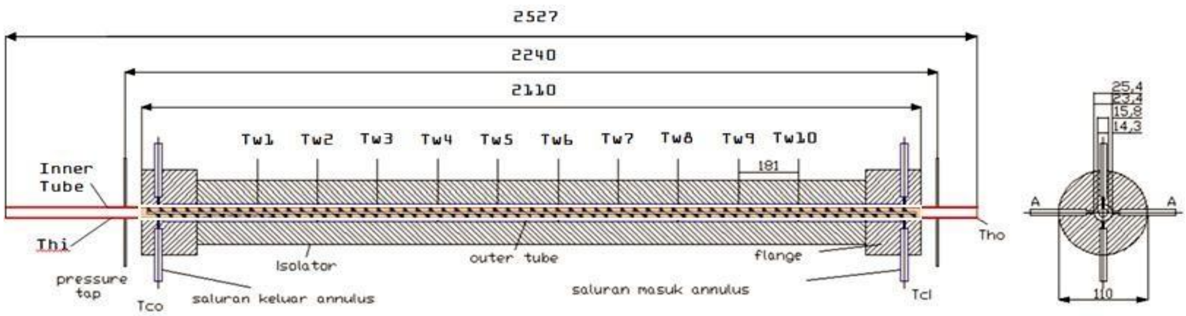

Figure 2. Concentric pipe heat exchanger scheme

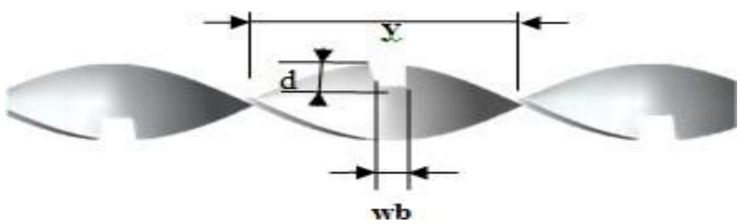

Figure 3. Length square-cut twisted tape insert nomenklatur on a pipe

Characteristic Calculation, Friction Factor, and Increasing Ratio of Heat Exchanger on Concentric Pipe

Heat transfer rate on innert tube can be stated as formula (1):

$$Q_h = \dot{m}.C_{p,h}.(T_{h,i} - T_{h,o}) = U_i A_i \Delta T_{LMTD} \quad (1)$$

Heat transfer rate to cold water on the annulus side could be defined as following formula (2):

$$Q_c = \dot{m}.C_{p,c}.(T_{c,o} - T_{c,i}) = h_o.A_o.(\bar{T}_{w,o} - T_{b,c}) \quad (2)$$

In which to define bulk temperature on concentric inner pipe as this following formula (3):

$$T_{b,c} = \frac{T_{c,i} + T_{c,o}}{2} \text{ dan } \bar{T}_{w,o} = \frac{\sum_{i=1}^{i=10} T_{w,i}}{10} \quad (3)$$

Heat loss percentage ($\%Q_{loss}$) could be defined as following formula (4):

$$\% Q_{loss} = \left[\frac{Q_h - Q_c}{Q_c}\right] \text{x } 100\% \quad (4)$$

Overall heat transfer coefficient based on surface area on innert tube which could be defined as formula (5):

$$U_i = \frac{Q_h}{A_i \ \Delta T_{LMTD}} \quad (5)$$

In which a counter flow heat exchanger ΔT_{LMTD} number could be defined as formula (6):

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \quad (6)$$

where $\Delta T_1 = (T_{h,i} - T_{c,o})$ and $\Delta T_2 = (T_{h,o} - T_{c,i})$.

Median of convection heat transfer coefficient on the annulus side could be defined by this following formula (7):

$$h_o = \frac{Q_c}{A_o \left(\bar{T}_{w,o} - T_{b,c}\right)} \quad (7)$$

Median of convection heat transfer coefficient on the innert tube side could be defined by this following formula (8):

$$\frac{1}{U_i \ A_i} = R_{total} \quad (8)$$

Then, to define innert tube overall heat exchanger by formula (9):

$$\frac{1}{U_i.A_1} = \frac{1}{h_i.A_i} + \frac{\ln(d_o/d_i)}{2\pi k_p L} + \frac{1}{h_o.A_o} \quad (9)$$

where

$$A_i = \pi.d_i.\ L$$
$$A_o = \pi.d_o.\ L$$

Therefore, median of convection heat transfer coefficient on the innert tube side could be defined as following formula (10):

$$h_i = \frac{1}{\left[\frac{1}{U_i} - \frac{d_i.\ln(d_o/d_i)}{2k_p} - \frac{d_i}{d_o.h_o}\right]} \quad (10)$$

An average Nusselt number on the innert tube side could be defined as this following formula (11):

$$Nu_i = \frac{h_i.d_i}{k_{fi}} \quad (11)$$

Heat exchanger effectiveness could be defined as following formula (12):

$$\varepsilon = \frac{Q_{aktual}}{Q_{maksimum}} = \frac{Q_h}{C_{\min}.(T_{h,i} - T_{c,i})} \quad (12)$$

pressure drop of innert tube could be defined as following formula (13):

$$\Delta P = \rho_m \ .\ g\ .\ \Delta h \quad (13)$$

Pumping power could be defined as following formula (14):

$$W_{pompa} = \dot{V}\ .\ \Delta P \quad (14)$$

Friction factor on the innert tube side could be defined as this following formula (15) berikut ini::

$$f = \frac{\Delta P}{\left(\frac{L_t}{d_i}\right)\left(\frac{\rho.V^2}{2}\right)} \quad (15)$$

Reynolds number of hot water flow on the innert tube side could be defined as this following formula (16):

$$\text{Re} = \frac{\rho.V.d_i}{\mu} \quad (16)$$

where of the hot water on innert tubes properties (ρ, kfi, dan μ) was evaluated by average bulk hot water temperature (Tb, h).

Heat transfer enhancement ratio factor at constant pumping power is the comparison of average convection heat transfer coefficient ratio of the innert tube in with a plain tube inserts which can be written by equation (17) below:

$$\eta = \left.\frac{h_s}{h_p}\right|_{pp} \quad (17)$$

Heat transfer characteristics, friction factor, and heat transfer enhancement ratio can be expressed in a row with a graph of Nu to Re, f to Re, and η to Re.

Heat transfer characteristics validation on insert pipe without plain tube inserts using (18) Distus - boelter and Gnielinski equation:

$$Nu = \frac{(f/8) \,.\, Re \,.\, Pr}{1{,}07 + 12{,}7 \,.\, (f/8)^{1/2} \,.\, (Pr^{2/3} - 1)} \qquad (18)$$

Petukhov formula (20) was applied for fully developed and number of $0.5 \le Pr \le 2000$ and $10^4 < Re < 5x10^6$.

Gnielinski equation(19):

$$Nu = \frac{(f/8) \,.\, (Re - 1000) \,.\, Pr}{1 + 12{,}7 \,.\, (f/8)^{1/2} \,.\, (Pr^{2/3} - 1)} \qquad (19)$$

Gnielinski equation (19) was applied for fully developed and number of $0.5 \le Pr \le 2000$ and $10^3 < Re < 5x10^6$. On (18) dan (19) equation, friction factor number (*f*) could be defined as following formula (20):

$$f = (0{,}790 \ln Re - 1{,}64)^{-2} \qquad (20)$$

Heat transfer characteristics validation on insert pipe without plain tube inserts as following Blasius equation (21):

$$f = 0{,}3164.Re^{-0{,}25} \qquad (21)$$

Blasius equation (21) for number of $4x10^3 < Re < 10^5$.

RESULT DAN DISCUSION

a. Plain Tube Validation

In this research, heat transfer characteristics validation for a plain tube was demonstrated by Gnielinski and Dittus- Boelter empirical correlations. Figure 4 below shows Nu_i to Re correlation on a plain tube.

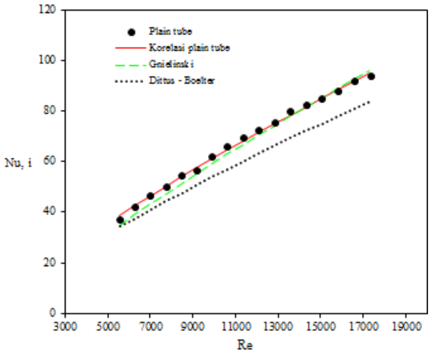

Figure 4. Nui to Re correlation on a plain tube graph

Figure 4 shows the average deviation of plain tube friction factor correlations to Dittus-boelter and Gnielinski, 13.8% and 3.4% respectively. Average deviation of Nui plain tube to Gnielinski and Dittus-boelter the correlations were less than 10% and 25%. ON actual plain tube average deviation to Dittus-Boelter correlation of 13.8%, Gnelienski by 3.4%, while the plain tube correlation of about 1.8%. There was an average deviation value which compared with Nu Gnielinski correlation, Dittus-Boelter correlation, and plain tube were still small enough, therefore the obtained data to Nu number on innert tube of the concentric pipe heat exchanger without twisted tape inserts, plain tube was valid. In this study also were validated friction factor characteristics of the plain tube by using Blasius empirical correlations. Characteristics of plain tube friction factor and innert tube friction factor (f) can be seen in Figure 5 below.

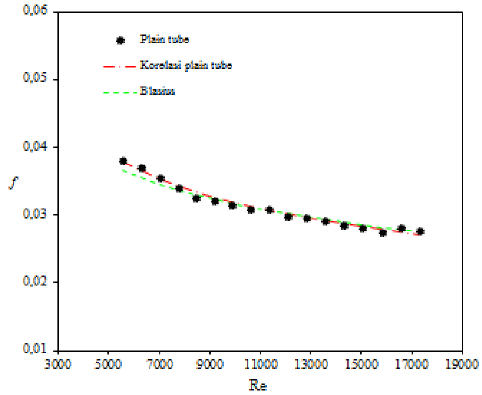

Figure 5. f to Re correlation on plain tube graph

Figure 4-5 shows a comparison between Nu_i and f to plain tube which compared to the empirical correlation calculation results. Deviations of plain tube Nusselt numbers against empirical Gnielinski and Dittus-consecutive boelter correlation were less than ±4.1% and ±13.4%. Whereas, deviation of the plain tube friction factor correlation was ± 1.8%.

b. Twist Ratio Effect to Heat Transfer of Heat Exchanger Characteristics with Square Cut Tape Insert and Classical Tape Insert.

Heat transfer characteristics of the heat exchanger tube in the concentric pipe can be seen in Figure 6.

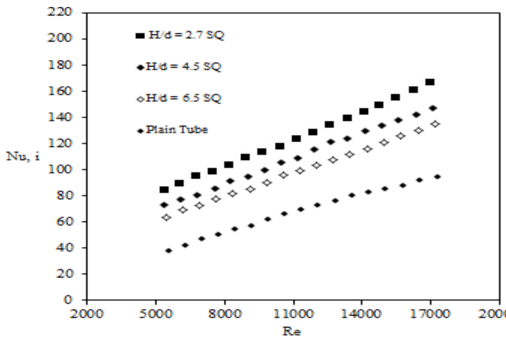

Figure 6. Nu_i to Re correlation graph

Figure 6 portrays that the larger Reynolds number, average Nusselt number (Nu_i) were increasing. It was met a plain tube or innert tube with square cut tape inserts and classical tape insert. Extra inserts square cut tape insert made Nu_i on innert tube larger than Nu_i plain tube. It proves that cut tape innert square inserts can increase the convection tube heat transfer rate. Extra square cut tape inserts caused the fluid flow turbulence intensity which passing through the tube wall was greater. It was generating the fluid mixing very well which resulting in an increased heat transfer (Murugesan, 2010).

The greater the twist ratio, pipe Nu_i will be decreased. It was caused by the greater twist ratio, to decrease the turbulence flow intensity on innert tube and also fluid dwell time becomes faster which causes Nui decreased. In Figure 6, the testing results showed that 5.400<Re<17.500, the average Nusselt number (Nu_i) on innert tube with square cut tape insert for twist ratio variations of 2.7; 4.5; and 6.5 successively increased in 79%; 58.2%; 43.5% which compared to the plain tube.

c. Twist Ratio Effect to Friction Factor Characteristic and Square Cut Tape Insert dan Classical Tape Insert Heat Exchanger

The pressure drop comparison characteristics between the plain tubes and square cut classical twisted twisted tape and tape were completed. In Figure 7 below shows Re to a pressure drop correlation at each variations.

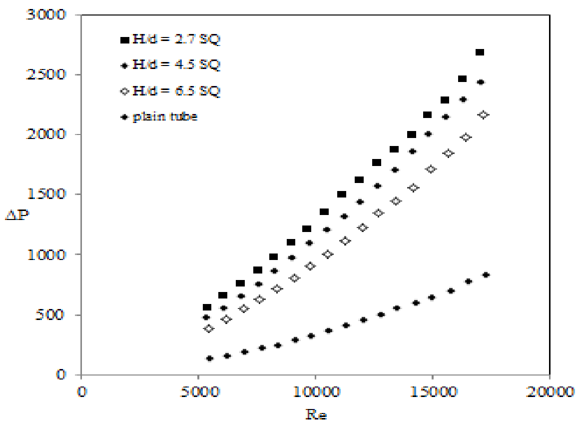

Figure 7. ΔP to Re correlation graph

Reynolds number of 5.400<Re<17,500 with a smaller twist ratio, the greater pressure drop on innert tube will occur. This could be originated due to the narrower twist ratio, the larger contact surface area and greater flow obstruction which causing a higher pressure loss. Average pressure drop on innert tube of inserts square inserts with cut tape twist ratio of 2.7; 4.5; and 6.5 were 2.41%; 1.96%, 1.44% respectively, greater than the pressure drop across the plain tube.

d. Twist Ratio Effect to Heat Exchanger Friction Factor Characteristic of *Square Cut Tape Insert* and *Classical Tape Insert.*

Friction factor characteristics with of square cut tape insert twist variations ratio of 2.7; 4.5; and 6.5 on innert tube of the concentric pipe heat exchanger can be seen in Figure 8. The larger the Reynolds number, the friction factor decreases both in plain tube or square cut tape insert. It was caused by the higher Reynolds number, the water flow rate on it will further increase as the friction factor was inversely proportional to its water flow squared velocity. Friction factor value on innert tube of square cut tape insert has a greater value than the plain tube. Furthermore, to recognize the friction factor to STT and CTT numbers correlation is shown in Figure 8 below.

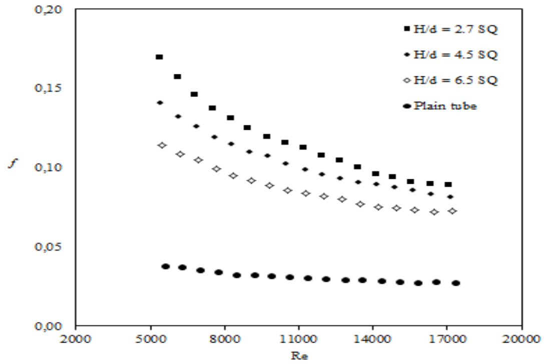

Figure 8. *f* to Re correlation graph

Figure 8 displays that the square cut tape insert with a smaller twist ratio, the greater friction factor. It may be caused by small twist ratio geometry and larger surface area to reduce the fluid free flow, which causing friction between the insert and the pipe wall getting larger (Suresh, 2012). In 5.300<Re<17.500 he friction factor was boosted on innert pipe of inserts square inserts with a cut tape twist ratio variations of 2.7; 4.5; and 6.5 were 2.58; 2.22; 1.73 respectively which is greater than the friction factor on a plain tube.

e. Twist Ratio Effect to Ratio Characteristic of Heat Transfer Enhancement of Heat Exchanger of Helical Screw Tape Insert

Extra inserts Square cut tape inserts provide variations in twist ratio affect the ratio of the heat exchanger with a pumping power enhancement. Figure 10 below shows the pumping power efficiency to Reynold number correlation.

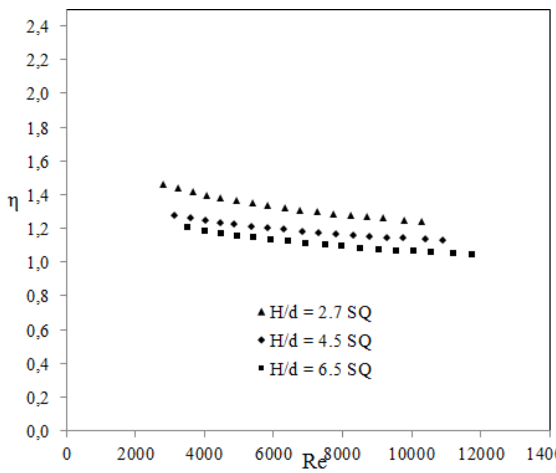

Figure 10. η dengan Re correlation graph

Figure 10 shows that the heat transfer enhancement ratio on innert tube which getting up to twist ratio was getting smaller due to the fluid turbulence flow which larger along with the lack of twist ratio. It portrays that the square cut tape insert with a twist ratio less storing energy capable of given operating condition. Figure 10 exhibits that adding effect of square inserts cut tape and innert tube into the heat transfer rate was a significantly improved.

Heat transfer enhancement ratio of heat exchanger of square cut tape inserts 2.7; 4.5; and 6.5 were 1.3; 1.2; 1.1 respectively. It means that at the same pumping power, convection heat transfer average coefficient value of the tube of square cut tape insert was greater than the convection heat transfer average coefficient on plain tube.It was appropriated to research (Murugesan, 2009).

CONCLUSION

Based on the testing results, analysis, and discussion by the previous chapter about the heat transfer and friction factor characteristics of the heat exchanger concentric pipe of square cute twist tape insert with 2.7, 4.5, and 6.5 of variations ratio can be concluded as follows:

1. The square cut tape inserts in 5500<Re<17500, has a Nu_i value, 79%; 58.2%; 43.5% respectively which compared to the plain tube. The friction factor of 2.58%; 2.22%; 1.73% and the heat transfer enhancement ratio of 1.3; 1.2; 1.1 times were larger than the plain tube.
2. The heat transfer characteristics, the friction factor, and a leverage ratio of twisted tape inserts heat was increased along the little twist ratio than without plain tube inserts.
3. A range of 5,500<Re<17,500 square cut tape insert has heat transfer characteristics value, friction factor, and heat transfer enhancement ratio was higher than the classical twisted tape.

NOMENCLATURE

A_c = channel surface area (m^2)
A_i = inner pipe surface area (m^2)
A_o = inner pipe external surface area (m^2)
$C_{p,c}$ = cold water spesific heat (kJ/kg.ºC)
$C_{p,h}$ = hot water spesific heat (kJ/kg.ºC)
d_i = inner pipe diameter
d_o = inner pipe external diameter
D = pipe diameter (m)
f = friction factor
f_p = plain tube friction factor
f_s = insertion pipe friction factor
g = gravity (m/s^2)
h_i = convection heat transfer coefficient inner pipe average ($W/m^2.ºC$)
h_o = convection heat transfer coefficient at annulus average ($W/m^2.ºC$)
h_p = convection heat transfer coefficient of average inner pipe without twisted tape insert ($W/m^2.ºC$)
h_s = convection heat transfer coefficient average inner pipe with inserts ($W/m^2.ºC$)
k_i = the average thermal conductivity of inner pipe hot water (W/m.ºC)
k_p = thermal conductivity of deep pipe material (W/m.ºC)
L_t = pressure drop measurement length of inner pipe (m)
$\dot{}$ = cold water mass flow rate (kg/s)
$\dot{}$ = hot water mass flow rate (kg/s)
Nu_i = average inner pipe Nusselt number
Nu_p = the average Nusselt number of plain tube
Nu_p = the average Nusselt number inner pipe using inserts
pp = equal pumping power
Pr = Prandtl number
Q_{aktual} = actual heat transfer rate of heat exchanger (W)
Q_c = annulus heat transfer rate (W)
Q_h = inner pipe heat transfer rate (W)
Q_{loss} = inner pipe heat loss convection (W)
$Q_{maksimum}$= maximum heat transfer rate Probably from a heat exchanger (W)
Re = Reynolds number
$T_{b,c}$ = average bulk temperature at annulus (ºC)
$T_{b,h}$ = average inner pipe bulk temperature (ºC)
$T_{c,in}$ = cold water temperatures enter the annulus (ºC)
$T_{c,out}$ = cold water temperature out annulus (ºC)
$T_{h,in}$ = hot water temperatures enter the deep pipe (ºC)
$T_{h,out}$ = hot water temperature out the deep pipe (ºC)
= average temperature of the outer wall of inner pipe (ºC)
U_i = overall heat transfer coefficient based on inner pipe surfaces ($W/m^2.ºC$)
U_o = overall heat transfer coefficient based on the outer surface of the deep pipe ($W/m^2.ºC$)
V = average speed of hot water at inner pipe(m/s)
Y = twist ratio
= inner pipe volumetric flow rate of hot water (m^3/s)
= pumping power (W)
ρ = hot water inner pipe density (kg/m^3)
η = heat transfer increased ratio
μ = inner pipe dynamic viscosity of hot water (kg/m.s)
Δh = height altitude fluid manometer (m)
ΔP = pressure drop (Pa)
ΔT_{LMTD} = the average logarithmic temperature difference (ºC)

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